

HIGH-SPEED LOW POWER RADIAL TURBOCOMPRESSOR FOR OIL-FREE HEATPUMPS

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ABSTRACT

The presence of oil in domestic heat pumps is a hindrance, particularly for enhanced surface evaporators and for advanced concepts based on two-stage cycles. Very compact oil-free dynamic compressor directly driven by high speed electric motors and supported on refrigerant vapor bearings represent a promising alternative. This paper presents a design analysis of the first stage prototype with the various tradeoffs between the impeller characteristics to follow the seasonable heat demand, the bearing and rotor stability and the constraints linked with the high speed electric motor. Refrigerant chosen for the first experimental approach is HFC 134a with a 20 mm diameter impeller, a 6 kW motor and self-acting bearings. The impeller design and predicted performance as well as the experimental facility for the initial tests are presented.

1 INTRODUCTION

Fear of global warming and the serious trend of oil and gas becoming sparse are strong incentives to reconsider ways of heating. Man discovered fire for heating and cooking more than one hundred thousand years ago. Today one puts a box around it, calling it a boiler, but is it a 21th century technology? Figure 1 compares a number of technological combinations to supply heating services, considering two different levels of supply temperature in the building. It typically shows the considerable amount of fuel saving which could be achieved in heating compared to today's predominant mode of heating technology, which is mainly simple boilers. Only heat pumps allow to get more useful thermal energy than the energy provided, because they upgrade heat from the environment. Of course these technologies do not need to be located at the same location as electricity from on cogeneration engine for example can be used to drive a heat pump in another building.

In several countries heat pump develop very successfully for new single houses equipped with low temperature heating like floor heating (corresponding to the 35°C front row in figure 1). However the largest market is the retrofit of oil or gas boilers in existing houses. Gradually efforts are being made to improve the building envelop but,

particularly in Europe, installed hydronic systems with radiators still require medium temperature heating up to 65° or 70°C (back row in fig 1). When considering air source heat pumps these requirements imply high temperature lifts. It is known that the exergy losses in single stage compression heat pumps are in the compressor (about 50%), in the expansion valve (about 35%) and in the heat exchangers (about 15%). Embedded in these values are cycling losses characteristic of on-off regulations. The present paper presents a concept aiming at reducing each of these losses:

- 1) Limitation of the cycling losses by using a variable speed compressor limiting on-off regulation to only one part of the season
- 2) Improved compressor efficiency with permanent magnet driving motor
- 3) Oil-free technology allowing an easier two-stage operation without having to compensate oil migration problems typical when using two separate stages of compression. Two stage cycles limit the expansion losses by dividing the expansion into two sub-expansions with interstage re-injection of the vapor produced downstream of the upper pressure valve.
- 4) Oil-free technology a) limiting pressure drops by eliminating the need for high vapor velocity at the end of the evaporator to carry over oil, b) reducing the higher superheating temperature required to degas refrigerant from oil c) avoiding oil clogging in the enhanced evaporator surfaces.

The efficiency gains linked to two stage operation have been demonstrated with separate scroll compressors in particular by Favrat et al. (1997). An intermediate solution to reduce expansion losses by using single stage scroll compressors with intermediate vapor injection ports was demonstrated by Zehnder et al. (2002, 2005). This can be called the two-stage cycle of the poor and is now gradually expanding in air based heat pumps in Europe! However the next efficiency step will be real two stage as intended by the study introduced in the present paper.

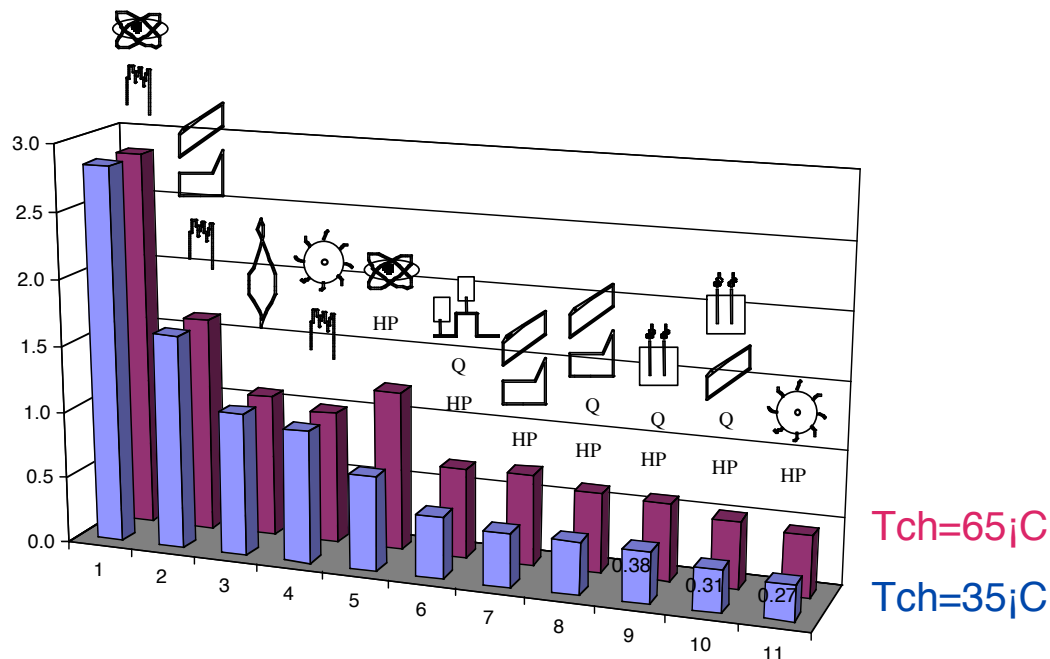


Figure 1: Comparison of various technology combinations for heating taking fuel boilers as a reference. (Front and back rows relate to the heating system supply temperature, with 1. Nuclear power and Joule heating, 2. Power from combined cycle and Joule heating, 3. Fuel boiler, 4. Hydropower and Joule heating, 5. Hydropower and heat pumps, 6. Cogeneration engine and heat pump 7. Power from combined cycle and heat pump, 8. Power from cogeneration combined cycle and heat pump, 9. Power from cogeneration fuel cell and heat pump, 10. Power from cogeneration hybrid fuel cell-gas turbine and heat pump, 11. Hydropower and heat pump)

Regarding low power oil free compressors let us cite, among other attempts, two trends which have been to modify volumetric compressors either by using rubbing seals in reciprocating compressors with relatively short maintenance intervals or by exploring liquid refrigerant lubrication in ball bearings of co-rotating scroll compressors (Molyneux et al. 1996, Morishita et al., 1988, 1996). This is possible in co-rotating scroll because, contrary to the orbiting scrolls, the two rotors are well balanced and the radial bearing loads are low. However two stage co-rotating scroll compressors would be more bulky to realize than with a concept of two impeller radial compressor on the same shaft, which is described underneath.

2 HEAT PUMP LAYOUT

2.1 Heat pump Cycles

The most important losses of high temperature lift heat pumps occur in the compression and the expansion processes; therefore it is of primary importance to develop advanced compressor design, which can take advantage of the cycles allowing a better recovery of the exergy of the liquid refrigerant at the outlet of the condenser. In a separate project Zehnder et al. (1998) analyses several of the most promising cycles often together with experimental validation. These cycles include:

1. Single stage cycle with an additional separate cycle for upgrading some of the subcooling heat of the condensed liquid to further heat the water to be heated.
2. Two superposed separate single stage heat pump cycles. A condenser-evaporator couples the two cycles by transferring heat from the bottoming cycle to the topping cycle.
3. Cycle with a two stage expansion, an economizer phase separator and (or) an internal heat exchanger at intermediate pressure and a single stage compressor with intermediate vapor injection (Beeton and Pham, 2003).
4. The same as 3 but replacing the single stage compression by a physical two-stage compressor

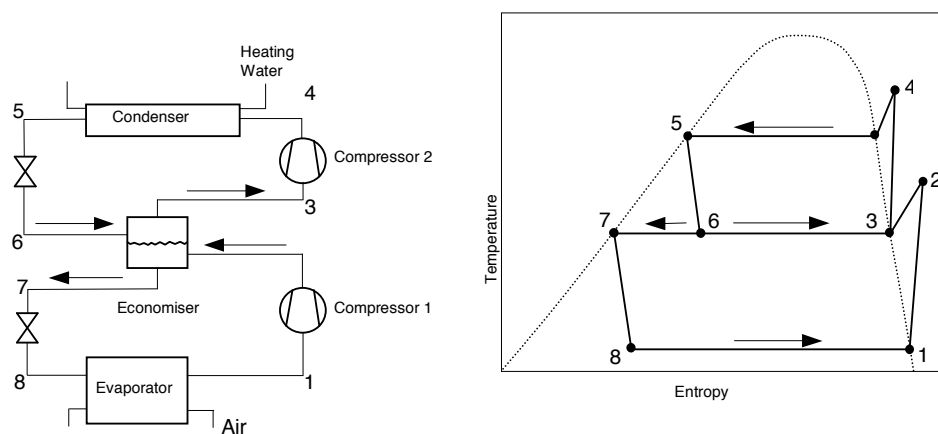


Figure 2: Flow chart and T-s diagram of a two stage heat pump with phase separator

The study has shown that the COP of solution 2 is the worst due to the internal heat transfer losses in the condenser-evaporator, whereas solutions 1 and 4 presented similar COPs. Solution 3 is an interesting short-term solution even if it provides lower COPs than solutions 1 and 4. Solution 1 is not interesting for radial compressors due to the high pressure ratio needed for the main cycle. Solution 3 and 4 with an internal heat exchanger but no economizer-separator are of particular interest for non-azeotropic refrigerant; the internal heat exchanger instead of the economizer avoids a distillation of the refrigerant-blend. However injection of a potentially two phase refrigerant

flow might be tricky to handle as high speed radial compressors will not cope with liquid droplets. Solution 4 in combination with an economizer-separator is clearly the most simple in terms of components and control. Furthermore it allows to partly defrost the evaporator by inverting the cycle and using the economizer as an internal energy source. The two stage cycle with an intermediate economizer-separator represented in figure 2 is the solution maintained in this project.

2.2 Refrigerant Choice

The criteria for the choice of the refrigerant used are of ecological nature (greenhouse effect, ozone decomposition) and of thermodynamic nature. The physical properties of the refrigerant define the pressure ratios, the mass and the volumetric flows and have a considerable influence on the COP. The choice of the refrigerant also influences the size and the rotational speed of the radial compressors. The analyzed refrigerants are R134a, R407c, and some natural refrigerants like butane (R600), iso-butane (R600a), propane (R290), and ammonia (R717). The synthetic refrigerants R134a and R407C do not have any ozone depletion potential but they have a large greenhouse effect if liberated to atmosphere. And the only alternative strategy is to swap to natural refrigerants that are either flammable or toxic. An interesting alternative would be CO₂, but it implies high pressures (100 bar), resulting in large forces on the bearings, and in very small compressor wheels.

The effects of the refrigerant choice have been calculated by Schiffmann et. al (2002). In order to estimate the rotational speed and the diameter of the two compressor wheels non-dimensional numbers have been used according to Balje (1981). The specific speeds have been chosen in order to fit the maximum efficiencies for radial compressors and the COP calculated for the cycle in figure 2 operating at the most demanding operation point in terms of pressure ratio. Based on this analysis refrigerant R134a was selected as the best fluid for this proof of concept application.

2.3 The Expected Heat Pump Performance

As shown by Schiffmann (2002) the annual COP of a two-stage heat pump as represented in Figure 2 equipped with radial compressors would be around 3.3 for a nominal heat rate of 12 kW at an external air temperature of -12°C and producing hot water at 60°C. This represents a significant increase compared to commercially available heat pump systems (single stage scroll compressor cycle with intermediate liquid injection).

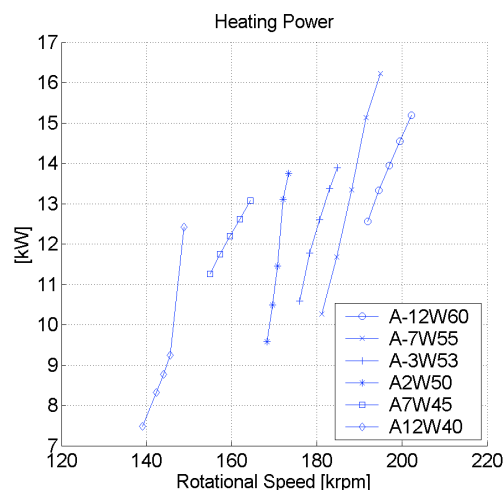


Figure 3: Heating power delivered by the heat pump as function of the rotational speed and of the operation point. "A-12W60" corresponds to an external air temperature of -12°C and a water supply temperature of 60°C

Figure 3 represents the heat rate delivered as a function of the rotational speed and of the operation points. The heat rate is maximum for low temperatures and minimum for higher air temperatures. This represents an inversion of the heating curve compared to conventional air-water heat pumps driven by constant speed rotary volumetric compressors. The radial two-stage compressor will therefore allow an operation with a much higher utilization coefficient, i.e. the heat pump will run in a much more continuous manner with less stop & starts than conventional systems.

3 COMPRESSOR DESIGN

The feasibility study by Schiffmann et al. (2002) has shown that the bearings running at high Mach numbers, the small impeller size and the power density of the electric motor represent high technological risks and are therefore important bottlenecks for the success of this system. It was therefore decided to build a single stage compressor serving as proof of concept before stepping directly to the two-stage compressor.

3.1 Impeller

The specifications for the compressor is summarized in table 1 representing 5 relevant operation points. The indicated mass flows correspond to minimum values required for a heat pump of adequate efficiency. A higher mass flow would lead to a higher instantaneous heat rate and might impose an intermittent use of the heat pump system.

Table 1: Specifications for the first compressor stage of a two-stage air-water heat pump

Operation Point		1	2	3	4	5
T_{AIR}	[°C]	-12	-7	2	7	12
P_{IN}	[MPa]	0.14	0.17	0.24	0.29	0.35
ρ_{IN}	[kg/m ³]	6.97	8.4	11.9	13.9	16.7
Π	[-]	4.2	3.4	2.4	2	1.7
Mass Flow	[kg/s]	$53 \cdot 10^{-3}$	$43 \cdot 10^{-3}$	$24 \cdot 10^{-3}$	$16 \cdot 10^{-3}$	$5 \cdot 10^{-3}$

The yearly overall efficiency of the heat pump corresponds to the ratio between the integrated heat rate and the consumed electrical power over the heating season. The heat pump compressor providing the highest yearly performance coefficient has to be primarily optimized (in terms of efficiency) for the most frequent operation point. The most frequent air temperature during the heating season in middle Europe is often of the order to 2°C. The impeller should therefore have a maximum efficiency at the pressure ratio of 2.4 and should nevertheless be capable of pressure ratios up to 4.2. The size of the impeller and the rotational speed should allow a safe operation in terms of rotor-dynamic stability; the size of the compressor should be at least such that it can be machined on a conventional 5 axis milling machine. This latter specification limits the minimum diameter and consequently also limits the maximum rotational speed. Based on scaling of known wheel geometries and investigations regarding complex surface machining it was decided to limit the minimum diameter to 20 mm.

A detailed description of the aerodynamic and of the mechanical impeller design is presented by Schiffmann and Favrat (2004). The impeller was developed using a 1D meanline analysis followed by a 2D inviscid code numerical study coupled with loss correlations for preliminary design, and the fine tuning of the blade geometry has been performed using the 3D Navier-Stokes solver (Concepts NREC software). Particular attention has been dedicated to the relative Mach number distributions and to the suppression of secondary flows. These latter are responsible for the jet-wake mechanism resulting in exit flow non-uniformity deteriorating the stage performance and limiting the operating range of the diffuser.

The resulting impeller has a diameter of 20 mm, with 9 main and 9 splitter blades, a backsweep angle of 50°, a channel exit height of 1 mm and an operating running speed between 110 and 220 krpm. The 15 µm tip clearance chosen (between impeller and shroud) is consistent with the use of gas bearings. Figure 4 represents the predicted compressor map for the main operation points, the predicted efficiency and the corresponding rotational speeds. The

maximum mass flow for the highest pressure ratio is slightly below the specifications. This results from a compromise between maximizing the mass flow at high pressure ratio and allowing continuous operation at the minimum mass flow corresponding to the point A2W50. As a result, the designed impeller allows a continuous operation for air temperatures lower than 2°C. At outside air temperatures exceeding 2°C an intermittent mode of operation will be necessary. The maximum impeller efficiency is expected to be in the range of 80%.

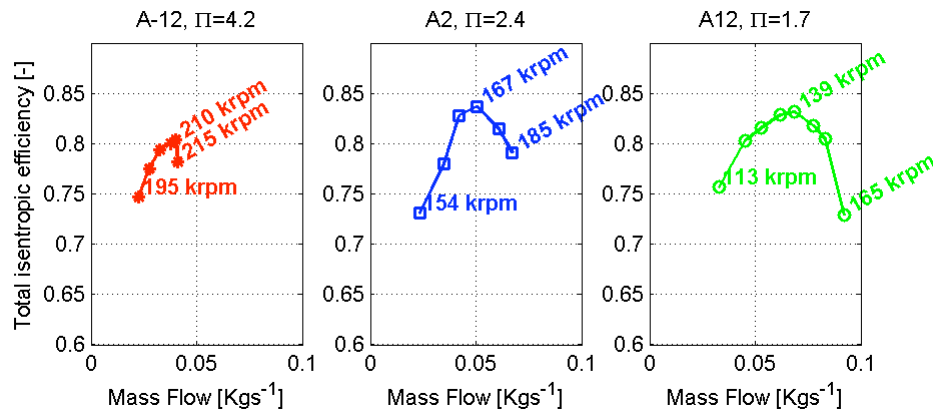


Figure 4: Compressor map for the three main operation points A12W60, A2W50 and A12W40 indicating the predicted efficiency and the rotational speeds.

3.2 Rotor

The final rotor will be composed of one (later two) radial compressor wheels (figure 5), an electric motor (figure 6), two dynamic radial gas bearings and two dynamic axial gas bearings. Several rotor layouts are possible presenting different advantages and inconvenients. The criteria for choosing the most suitable layout version include a first critical bending speed sufficiently above the maximum operating speed, easiness for production and assembly as well as cost minimization.

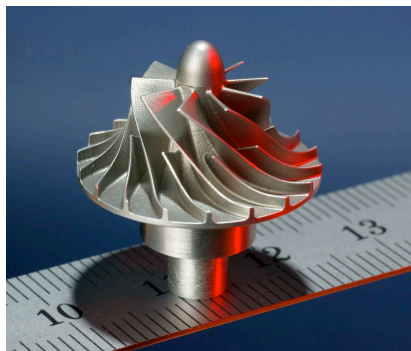


Figure 5: The machined impeller



Figure 6: The electric motor

The bearings used in this application are dynamic gas bearings. This type of bearings presents the advantage of not requiring the need for oil lubrication and therefore allowing very high rotational speeds with low mechanical losses. By their nature they present low damping coefficients and are highly cross coupled. Therefore their design is an integral part of the complete rotor design in terms of rotor dynamic stability. As the bearings will operate in vapor refrigerant close to the saturation curve, real gas effects need to be taken into account. The existing theory (based on the assumption of perfect gas) has been modified in order to take into account the real and the rarefied gas effects. This work is presented more into detail by Schiffmann and Favrat (2006). A model of gas bearing supported rotors

has been generated and the bearing design together with the rotor optimized for maximum stability and minimum power losses using an evolutionary Algorithm, which has been developed by Leyland (2002) and by Molyneux (2002).

4 TESTRIG DESIGN

The aim of the testrig is to allow the measurement of the turbocompressor performance over the complete operating range. It is going to operate in a closed loop with refrigerant R134a. Two possible rigs could be envisaged: A real single stage heat pump cycle with a condenser and an evaporator where the compressor inlet and exhaust pressures are regulated through the temperature levels of the cooling and the heating circuit. The second possibility would be a vapor circuit, where the test cycle operates only on the right side of the saturation line, i.e. vapor is compressed, expanded, cooled and led to the compressor inlet again. Due to a shorter response time to temperature commands the authors have decided to design a vapor test rig as such a rig would allow shorter test sequences.

Figure 5 represents a schematic view of the test rig as designed and built. The main refrigerant circuit is composed of the compression unit, a mass-flow meter, a controllable expansion valve, a cooling heat exchanger and a separator. The heat exchanger dissipates the heat generated by the compressor through an external cooling system. Liquid and vapor refrigerant will be present in the heat exchanger; the liquid will stabilize the pressure of the vapor sucked by the compressor. The bearing gas circuit is composed of a pressure expansion valve for stabilizing the ambient bearing pressure, of a filter avoiding contamination of the bearings and of a micro-valve allowing the regulation of the cooling mass flow. The gas is bled from the compressor exhaust through the bearings into the low pressure zone of the main refrigerant circuit.

The instrumentation of the test rig allows the acquisition of all relevant data for the determination of the compressor performance and for real time monitoring of the bearings, of the electric motor and of the operation of the complete testing loop. Two separate acquisition cards are being used: one for rapid sampling of the bearing position as well as compressor and bearing pressures, and a slower one for temperature acquisition. The two boards are piloted through a VI generated on LabView.

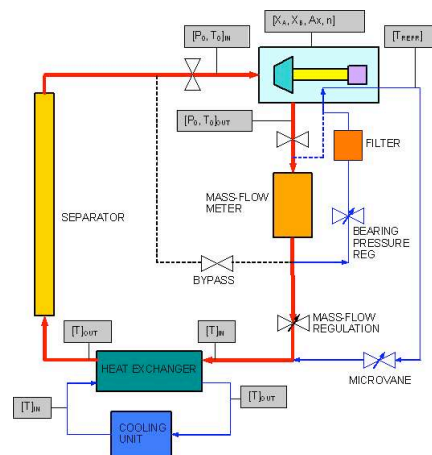


Figure 7: Schematic view of the test rig illustrating the main cycle and the motor cooling cycle

5 CONCLUSIONS

The procedure of the design process of a proof of concept high-speed low power radial compressor has been described from the specifications to the final design. The unit is aimed at the application of domestic high temperature lift heat pumps. Some of the elements composing the unit are already available and testing should start in the near future.

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ACKNOWLEDGEMENT

The authors acknowledge the financial contribution of the Swiss Office of Energy for the first part of the project and would like to thank the Fischer Group for their support of this work and for the permission to publish this paper.